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INTRODUCTION

A Roots blower is a form of positive displacement compressor and usually has two rotors mounted on parallel shafts and rotating in opposite directions within a sealed casing. There are clearances between the rotors (LOBE) and between the rotors and the casing (TIP and END plate) so that no internal lubrication is necessary and the air delivered is oil-free. Wear does not occur and therefore volumetric efficiency does not alter during the life of the blower. However, volumetric efficiency does not achieve theoretical maximum values due to leakage flow through the internal clearances, from the high pressure delivery side of the rotors to the low pressure inlet side (Fig.1). There is also leakage along the rotor shafts which are driven by external precision gears.

As the rotors revolve, air is drawn into the space between the rotors and casing, where it is trapped as the tip of the rotor passes the edge of the inlet opening. As the rotation continues, the opposite tip of the rotor passes the edge of the outlet opening (Fig.1) and the trapped cell volume of air is pushed through the outlet into the air line. Thus the air is only displaced by the blower since no volume change takes place as the charge is transferred from inlet to outlet. Actual compression takes place only when the tip of the leading rotor lobe has passed the edge of the delivery port and allowed a back flow or reflux of high-pressure air to resist the delivery of the initial charge in the cell volume. This compression process is essentially irreversible and may be accompanied by significant wave action; together with leakage flow this means that the volumetric efficiency of a Roots blower will be appreciably less than unity.

Review

Attempts have been made to estimate the amount of internal leakage flow and, in general, accurate results have not been achieved without resorting to empirical methods. Schopper (1) considered a blower with rotors of equal length and diameter, and calculated the leakage flow from the clearance area, the pressure difference and an empirical coefficient which was a function of the pressure difference. Winter (2) considered the total leakage as flow through a nozzle and calculated the flow using the clearance area as the nozzle throat area. The flow was assumed to be subsonic below an overall pressure ratio of 1.9 and sonic above it. Comparison of calculated and experimental values of volumetric efficiency for pressure ratios up to 1.7, showed that leakage was over-estimated below a certain speed and underestimated above this speed. Leakage was also underestimated when sonic conditions were assumed. Ertl (3) also used this approach but tried to compensate for these errors by introducing a throughput factor, i.e. an empirical coefficient. Patterson and Ritchie (4,5) have attempted an analytical treatment of Roots blower performance in general and of geometry and leakage aspects of involute rotors in particular. They claimed a prediction of leakage in terms of blower dimensions and of variations in pressure ratio and speed. However, in the calculation, empirical relations were used for friction and heat transfer characteristics. The leakage flow paths were considered to be two rectangular channels - corresponding to END and TIP clearances. The LOBE clearance leakage flow was incorporated with that of the TIP clearance although the flow could not reasonably be considered to pass through a parallel-sided channel - an assumption which could be valid for clearances at 'landed' rotor tips. Poiseuille conditions of fully-developed, laminar incompressible flow were assumed for flow through the clearance channels; the iteration procedures required for evaluation of the derived equations hardly seemed justified by the shortcomings of the formulation.

The accurate estimation of leakage flow is of fundamental importance for design and development of Roots blowers and for prediction of blower performance. This paper describes the results of an investigation which formed part of a comprehensive programme of research into different aspects of the performance of Roots blowers. The experimental part of this investigation sets out to identify and measure the leakage flow through three different types of internal clearance. For this purpose high pressure air was passed through a modified blower with stationary rotors and the values obtained were compared with theoretical values. When air passes through a Roots blower it is heated during reflux compression and the leakage air thus adds heat energy to the inlet air with which it mixes (Fig.2). Heat energy is also transferred from the delivery air to the rotors and casing, from the rotors and casing to the inlet air, and from the outside of the casing to the atmosphere. An analysis has been made of the effect of this heat energy transfer on the rotor, casing and air temperatures, and the resulting effect on the clearances and volumetric efficiency. The values of equilibrium temperatures and heat transfer coefficients are used to find the temperature distribution around the rotors and casing. Comparisons are made with results of dynamic tests with a Roots blower similar to that used for static tests.

THEORETICAL CONSIDERATIONS

The geometry of Roots blower rotors is of basic importance for the theoretical quantity of air delivered and the corresponding volumetric efficiency. This paper considers the Roots blower with two-lobed involute rotors and the theoretical rotor profile is designed for no clearance between the rotors. There should be only one point of contact for a given position of the rotor and the locus of this point of contact should be continuous. If these conditions are not fulfilled, then air will be trapped between the rotors and transferred from the high pressure to the low pressure side of the blower. The air delivered from the cell volume (between the rotor and casing) to the high pressure side is at low pressure (temperature) conditions, whereas the trapped (or 'carry-over') volume returned to the low pressure side is at high pressure (temperature) conditions. In effect, this increases the 'blockage factor' of the rotor. The blockage factor is a convenient non-dimensional parameter which can be used in Roots blower geometry analysis. It is defined as the ratio of the rotor cross-sectional area to the area of a circle with diameter equal to the rotor diameter and the volume of air delivered per cycle is proportional to (1 - blockage factor). Thus for maximum delivery the blockage factor should be as low as possible.

The analysis of Roots blower geometry develops a basic analysis by Castle (6) for involute (and cycloid) rotor profiles. In the calculation of the area for END leakage between the ends of the rotors and the casing, the length (ℓ) varies with the position angle of rotation, β. This length is assumed to be normal to the direction of leakage flow (Fig.1). The length was first approximated to two straight lines, i.e. ($T_1S + ST_2$); then to four straight lines, i.e. ($T_1W_1 + W_1S + SW_2 + W_2T_2$). Assuming that the inlet and outlet ports start in the same plane as the rotor centres, then there is a point of discontinuity at $\beta = 0$ i.e. when the rotors are mutually perpendicular. For this position the length is ($T_3S + SW_2 + W_2T_2$) and would also be valid for a casing with cusps. These lengths divided by the rotor diameter were computed for different values of β.

For determination of the leakage characteristics of the modified blower with static rotors for different values of β the high pressure side is effectively the 'inlet' side and the low pressure side is the 'outlet' side for the air supply to the stationary blower. Dimensional analysis (based on Huntley's method (7) of vector dimensions) produces a relationship between a non-dimensional leakage mass flow and the relevant independent variables, of the form:-

$$\frac{W\sqrt{T_o}}{p_o d^2} = f\left[\frac{C_E}{L}, \frac{C_T}{r_p}, \frac{C_L}{r_p}, \frac{d}{L}, \frac{\rho L u}{\mu}, \frac{\Delta p}{\rho u^2}, \frac{p_h}{\rho u^2}\right]$$

$$= f\left[\frac{C_E}{L}, \frac{C_T}{r_p}, \frac{C_L}{r_p}, \frac{d}{L}, (Re_d \times \frac{L}{d}), \frac{p_h}{p_\ell}, \frac{p_h}{\rho u^2}\right]$$

where C_E is rotor END plate clearance

C_T is rotor TIP clearance

C_L is internal LOBE clearance

d is diameter of inlet pipe

L is rotor length

ℓ is end plate leakage length ($\equiv r_p$)

p_h is upstream (high) pressure in inlet pipe

p_ℓ is downstream (low pressure) in outlet pipe

p_o is total head pressure (inlet)

Δp is pressure drop across blower

Re_d is Reynolds number for flow in inlet pipe

r_p is pitch circle radius

T_o is total head temperature

u is flow speed in inlet pipe

W is leakage mass flow

ρ is upstream density of air

μ is viscosity of air

θ is pressure ratio - p_h/p_ℓ

Huntley's method gives significance to the geometry parameters C_E/L and d/L; the former provides a distinction for the performance of 'long' or 'short' machines with geometrically similar profiles which retain practical production clearances (C_E, C_T, C_L); the latter allows for design changes in inlet pipe/manifold configurations.

A detailed theoretical process analysis of Roots blower performance was developed by Cole, Groves and Imrie (8) and non-dimensional relations derived on the basis of the following assumptions:-

(1) The working fluid is a perfect gas - ratio of specific heats, γ
(2) All processes are adiabatic
(3) Mixing processes proceed instantaneously to homogeneous equilibrium
(4) The leakage flow is considered isentropic to the throat of an equivalent area convergent nozzle. The subsequent throttling and mixing processes produce a rise of intake temperature.
(5) The delivery space acts as an infinite receiver and there is no significant effect due to wave-motion.

The isentropic efficiency of the perfect blower (or Roots or type efficiency (3) (9)) is given by

$$\eta_{is} = \frac{\text{isentropic work}}{\text{Roots work}} = \frac{\gamma}{\gamma - 1} \cdot \frac{\theta^{(\gamma-1)/\gamma} - 1}{\theta - 1}$$

The effect of leakage flows is to introduce a volumetric efficiency term ($\eta_v \times \eta_{is}$). A leakless Roots blower can only achieve higher pressure ratios at the expense of a decrease in overall efficiency; if leakage paths are present throughput (of air) will decrease as the pressure ratio rises, and a further loss will follow (η_v).

EXPERIMENTAL WORK

A standard Roots blower, 5 inches long by 5 inch centres (8.1 inch tip dia.) was modified for static leakage tests. The metal headplate was replaced by a perspex copy so that the positions of the rotors could be examined easily. The gears were left exposed and a pin used to lock the gears (40 teeth - 9° increments in angular displacement β) and therefore the rotors in position. Shims were used to vary the end plate clearances which were divided equally between the two ends by setscrew adjustment. Lobe/lobe and tip/casing clearances were also measured, for each rotor position, with feeler gauges. All clearances were accessible from either the high pressure or low pressure side and each clearance was measured two or three times along the length of the blower and an average value taken.

The principal tests carried out measured:-

(a) leakage through all internal clearances, with end clearances of 0.016, 0.021 and 0.027 inches equally divided between the two ends;

(b) leakage through internal clearances with lobe/lobe clearances blocked and end clearance of 0.021 inches equally divided;

(c) leakage along the shafts with an end clearance of 0.021 inches equally divided.

Compressed air was supplied through 2 in dia. copper tube to the test blower and controlled by a pressure regulator and valve upstream and by a valve downstream of the blower, to regulate the pressure ratio across the blower. The internal 'leakage' air was then measured using an orifice plate and corner tappings to BS 1042:1964 (10) specifications. Pressure measurements (p_h) upstream and (p_ℓ) downstream of the blower were made by manometers at tappings in the 2 in dia. tubes 2 feet from the ports. Lobe/lobe clearances were blocked effectively by rolling a thin sheet of rubber between the lobes of the rotors.

Results

Tests (a) and (b) were used together to estimate the leakage through the different clearances for different angular positions and, with (non-dimensional) leakage = $(1 - \eta_v)$, values of volumetric efficiency (η_v) were obtained for pressure ratios θ = 1.2, 1.3, 1.4, 1.45, 1.5 and 1.6. A typical set of results is shown in Fig.3 for θ = 1.6, extrapolated to zero end plate clearance for each position of the rotors. The difference between the extrapolation value of volumetric efficiency and the actual volumetric efficiency, can be ascribed to END clearance leakage. The difference between the values of η_v for an end clearance of 0.021 in and for the case with the lobes blocked is a measure of the LOBE leakage. Average values for all angular positions of the rotors were calculated.

Table of average values of leakage, $L = (1 - \eta_v)$

Pressure ratio, θ =	1·2	1·3	1·4	1·45	1·5	1·6
(m) ΣL-16	0.325	0.406	0.470	0.499	0.528	0.583
(m) ΣL-21	0.407	0.506	0.588	0.626	0.661	0.729
(m) ΣL-27	0.468	0.580	0.676	0.721	0.765	0.849
(m) L-T21+L-E21	0.294	0.373	0.439	0.470	0.498	0.554
(e) L-T0+L-E0	0.125	0.162	0.182	0.189	0.196	0.207
L-E16	0.200	0.244	0.288	0.310	0.332	0.376
L-E27	0.343	0.418	0.494	0.532	0.569	0.642
L-E21	0.282	0.344	0.407	0.437	0.465	0.522
L-L21	0.113	0.133	0.149	0.156	0.163	0.175
L-T21	0.012	0.029	0.032	0.032	0.034	0.032

(m) - measured; (e) - extrapolated; 16 ≡ 0.016 in etc. (C_E); ΣL - leakage through all clearances; E - END; L - LOBE; T - TIP. The leakage flow along the shafts did not vary much with angular position of the rotors.
For θ = 1.3, shaft leakage = 0.041 ± 0.007
For θ = 1.45, shaft leakage = 0.057 ± 0.010

The shape of the graph in Fig.3, can be ascribed almost completely to variation in lobe leakage, which was caused by variation in lobe clearance with angular position. The 'front' clearances are those between the leading faces of the driving rotor and the trailing faces of the driven rotor. They are greater than the 'back' clearances and therefore the leakage for positions 1-9, 21-29 is greater than for positions 11-19, 31-39 (Fig.3).

Log-log plots established the following average empirical results: END leakage (60-70%) is proportional to rotor end plate clearance and proportional to $(\theta - 1)^{0 \cdot 6}/\theta$;
LOBE leakage (20-30%) is proportional to lobe/lobe clearance and proportional to $(\theta - 1)^{0 \cdot 4}/\theta$;
TIP leakage and SHAFT leakage are small (5 - 10%) in comparison.

Comparison with dynamic test results

Another (5 x 5) blower was used for dynamic tests (11) and the average cold clearances were (static test blower in brackets): 0.007 in (0.009) tip/casing; 0.008 in (0.008) end/casing; 0.013 in (0.012) lobe/lobe. For the dynamic tests rotor tip velocity is not considered to be significant; 'cold' clearances change due to thermal effects; low (inlet) pressure is usually constant at atmospheric pressure - in the static tests the high pressure was kept constant. This accounts for the difference (proportional to $(\theta - 1)$) between the dynamic and static volumetric efficiencies for a speed of 1000 rev/mm. The dashed line shows the reasonable correspondence obtained when the static leakage flows are standardised by dividing by the pressure ratio, θ. Cole, Groves and Imrie (8) report a similar correspondence for dynamic and static test results carried out on a different blower (6 in long x 6.1 in tip diameter) for speeds of 1400, 2000 and 3000 rev/mm. 'Static' volumetric efficiencies are sufficiently accurate for pressure ratios up to about 1.5; but above this, increasing account must be taken of the changes in leakage area owing to differential-expansion effects.

For comparison, leakage was also calculated using Winter's method (2) for conditions similar to the static leakage tests:

(Pressure ratio, $\theta = 1.45$)	TIP	LOBE	END
Clearances (inches)	0.009	0.012	0.016
Mean area (in^2)	0.0920	0.0585	0.2004
Winter			
Throat velocity (ft/s)	796	796	796
Non-dimensional leakage	0.181	0.115	0.394
Experimental			
Non-dimensional leakage	0.032	0.156	0.310
Throat velocity (ft/s)	141	1081	672
Reynolds number (clearance)	$0(10^3)$	$0(10^4)$	$0(5 x 10^3)$

Winter's method assumes that all clearances are equally important and is reasonably effective for predicting overall leakage but could be improved by an empirical flow coefficient. Nozzle 'irreversibility' produced a value of 1.15 for the polytropic index, n, for p/ρ^n = constant. Laminar incompressible flow is not a realistic assumption for END plate leakage flow.

ROOTS BLOWER CYCLE ANALYSIS WITH HEAT TRANSFER

This analysis deals with the effect of heat energy transfer from the high pressure air to the rotors and casing, and from the leakage air to the inlet air. The rise in temperature of the rotors and of the casing cause changes in internal clearances and thus in volumetric efficiency. Consider the (n)th revolution of a Roots blower at speed N rev/m and pressure ratio θ (Fig.4): mass m_n of air taken in at (T_i, P_i) mixes with leakage mass $x_{n-1}\, a_{n-1}\, m_i$ at (T_{ddn-1}, p_d) to give mass $a_n m_i$ at (T_{in}, p_{in}). Heat (H_{rin}) is lost from the rotors at T_{rn-1} and (H_{cin}) from the casing, to the inlet air to give rotor temperature T_{rin}, and air conditions (T_{iin}, p_{iin}). Air is conveyed to the delivery side and compressed to (T_{dn}, p_d). Heat is transferred to the rotors and casing, and heat is lost from the casing to the atmosphere, i.e. rotor and casing temperatures rise from T_{rin} to T_{rn}, T_{cn-1} to T_{cn} respectively. Air temperature drops from T_{dn} to T_{ddn}. Clearances change with temperature from C_{n-1} to C_n, altering the volumetric efficiency from $(1 - x_{n-1})$ to $(1 - x_n)$. Mass $(1 - x_n)\, a_n m_i$ of air is delivered at (T_{ddn}, p_d). Leakage mass is $x_n a_n m_i$. Simplifying assumptions included constant inlet and outlet conditions, average values of parameters used for each revolution, instantaneous mixing of leakage air and temperature gradients around the rotors and casing and across the casing were neglected.

An iterative process was used to determine blower conditions after each revolution and the computer program was run to 1.5×10^5 revolutions for a 5 x 5 blower with end clearance = 0.008 inches, lobe clearance = 0.012 inches, atmospheric pressure = 14.7 lbf/in^2 and atmospheric temperature = 20°C, for (a) $\theta = 1.6$, N = 1500 rev/min (b) $\theta = 2.0$, N = 2000 rev/min. A graph of temperature of delivered air, rotors and casing is shown in Fig.5. The analysis was developed by the lumped parameter network technique as described by Shillabeer, Thomas and MacInnes (12) to give the temperature distribution around the rotors and casing. The maximum difference between the trailing and leading portions of the rotor was almost 10°C. Temperature indicating paints used on the rotors of a 5 x 5 blower running normally showed a temperature difference of approximately 5°C between the trailing and leading portions of the rotor (11).

COMMENT

Further development of experimental techniques for evaluating leakage flows through internal clearances under static conditions would provide more insight into the mechanisms of leakage flow together with improved empirical measurements of coefficients and indices. This information would enable more effective models of dynamic performance to be established thereby improving the design and development of Roots blowers.

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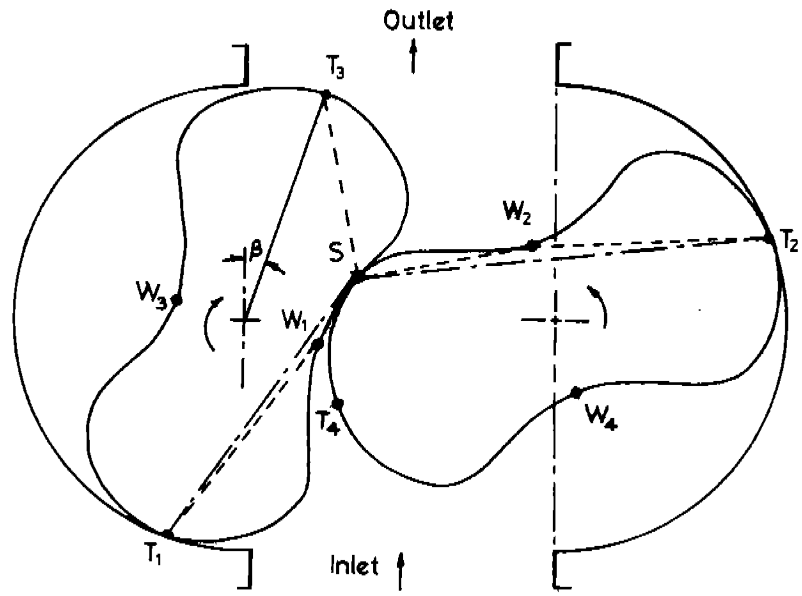

Fig.1 End plate clearance leakage lengths.

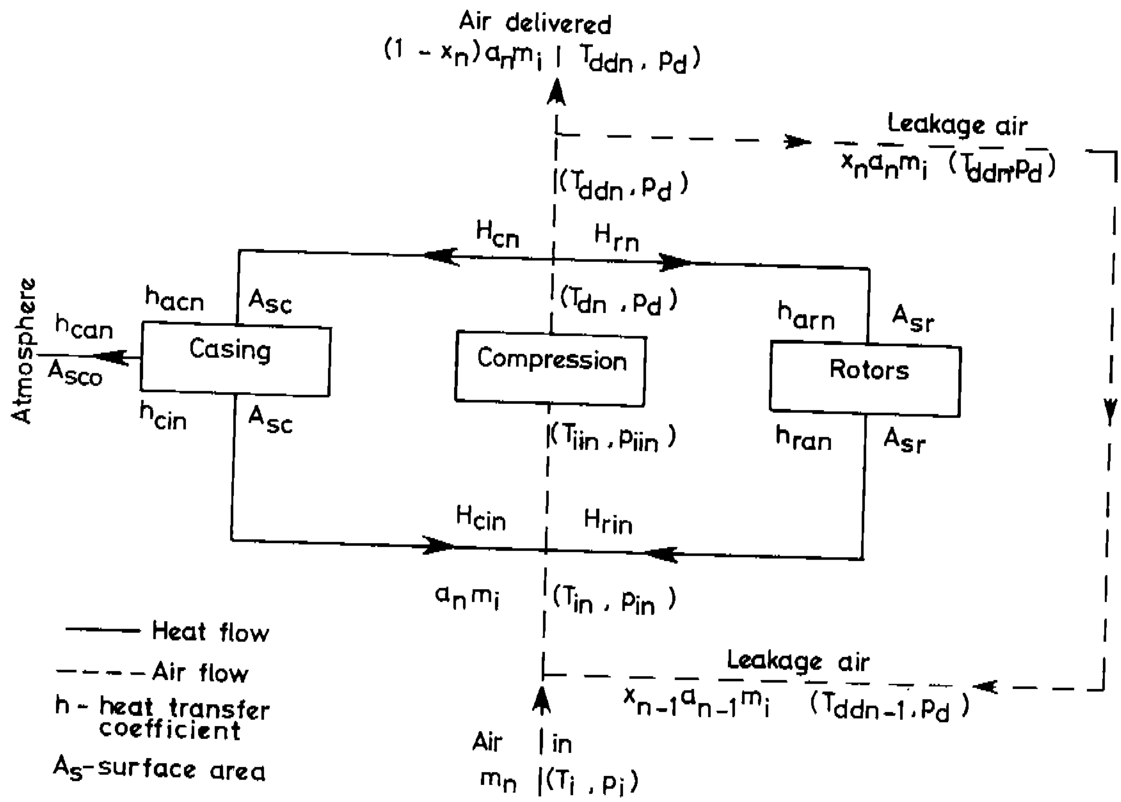

Fig.2 Diagram showing heat and air flows for the n^{th} revolution

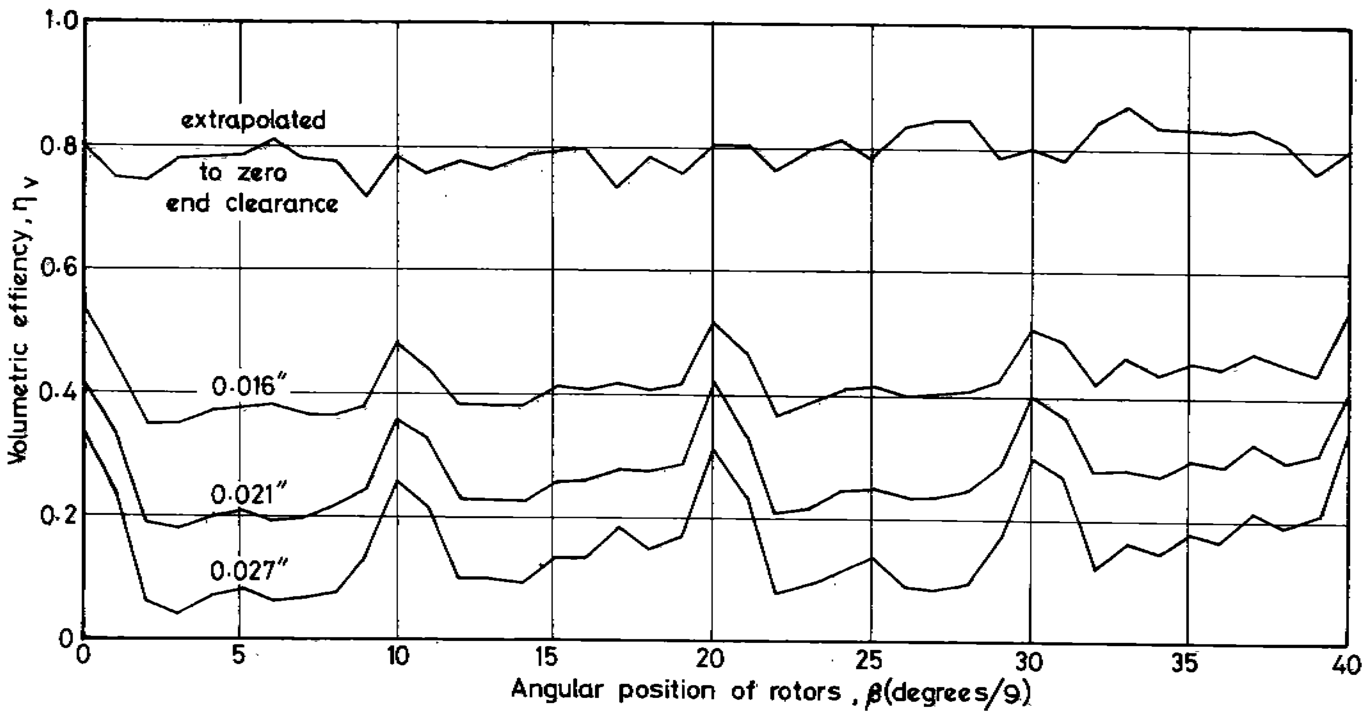

Fig. 3 Variation of η_v with β and end clearance - pressure ratio, $\Theta = 1.6$

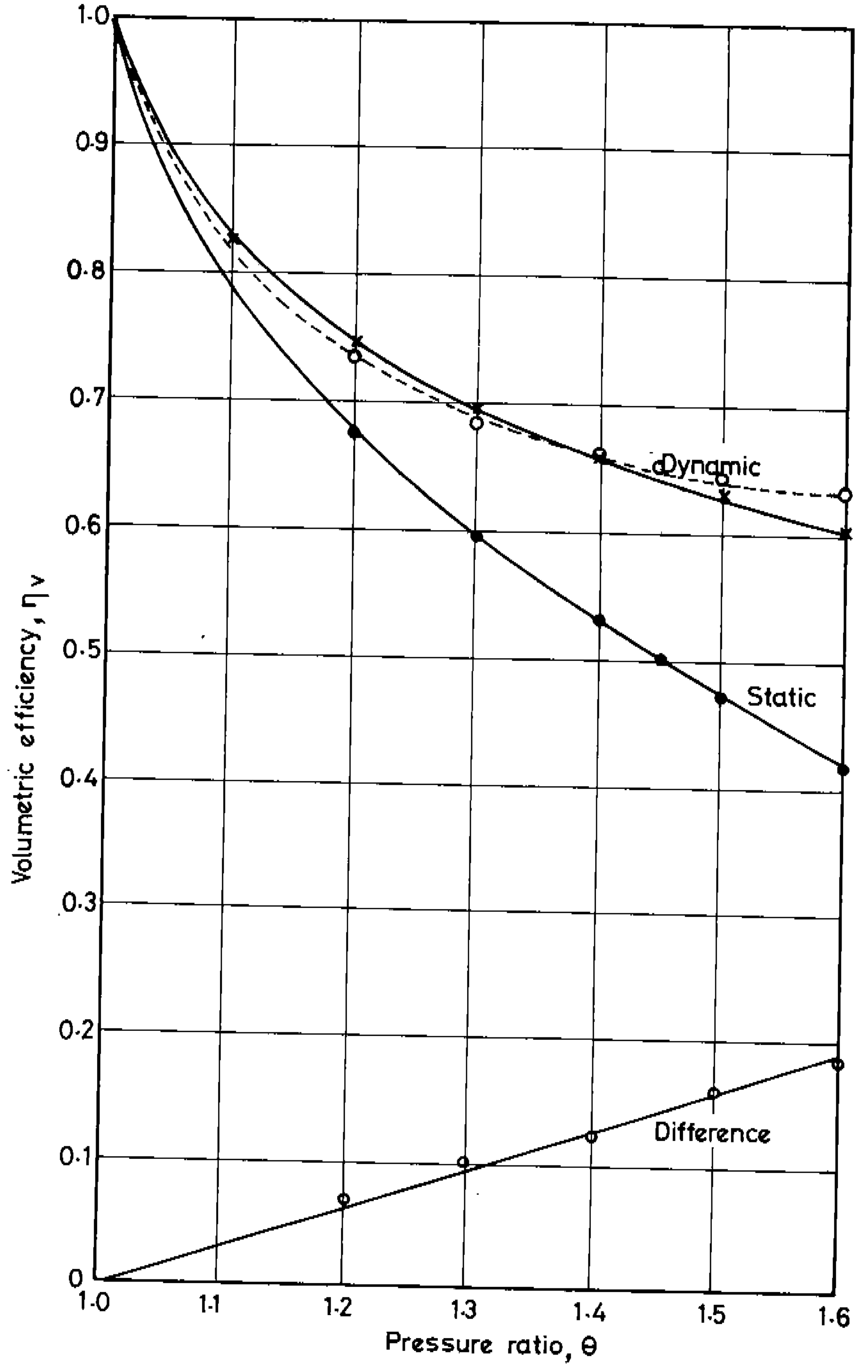

Fig. 4 Comparison of static and dynamic test results

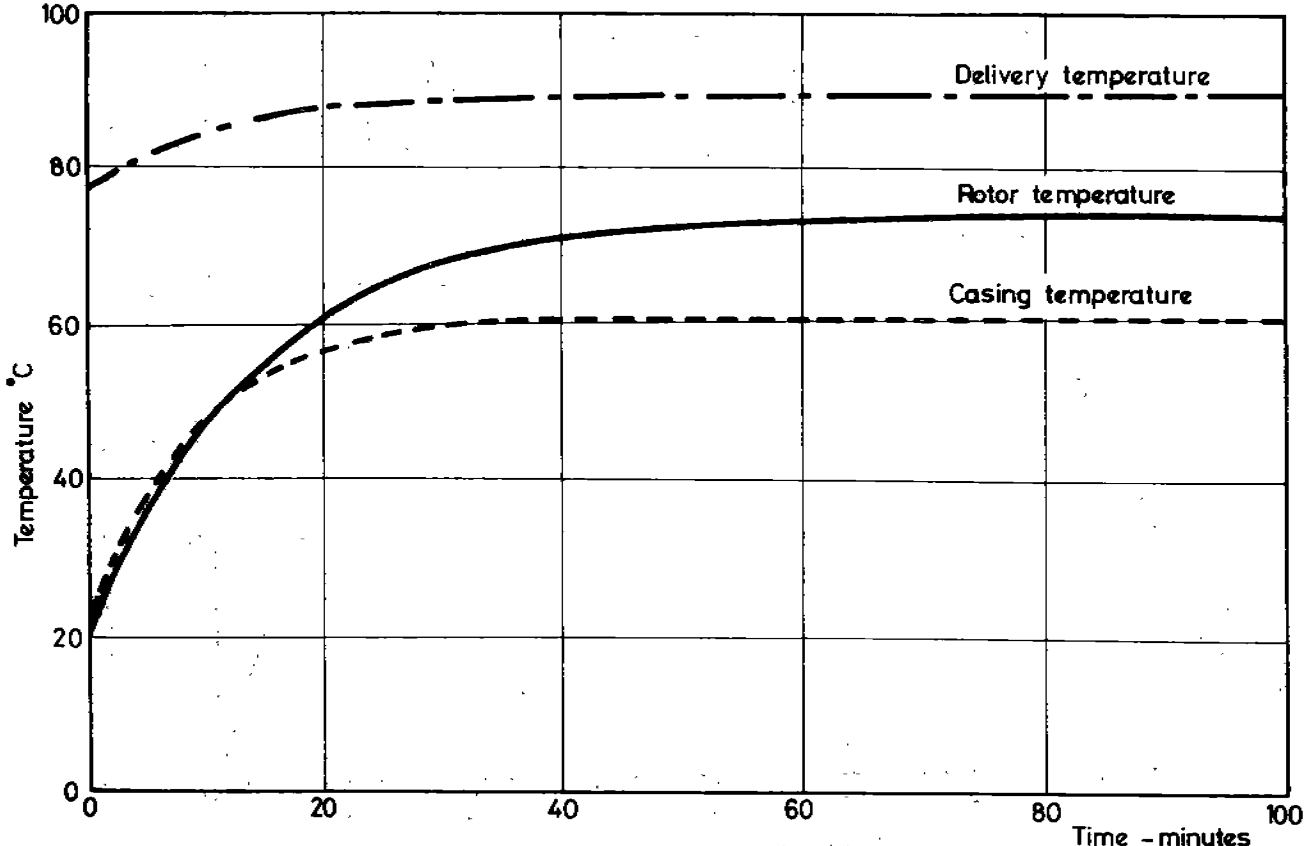

Fig. 5 Variation of temperature with time. Pressure ratio, $\Theta = 1{\cdot}6$, Speed = 1500 rev/min